PARTICULAR PROBLEMS OF STEAM TURBINE LUBRICATION

by

Walter E. Enz

Manager, Turbine Controls

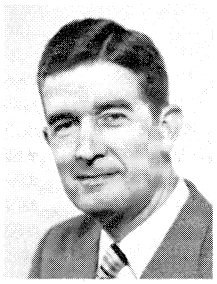
and

Alfred Häusermann

Chief Turbine Design Engineer

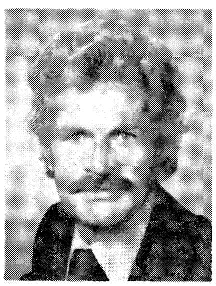
BBC Brown, Boveri & Company, Limited

Baden, Switzerland



Walter E. Enz, a native of Switzerland, was graduated as a mechanical engineer from the Institute of Technology at Winterthur. After three years of design practice in France and England, he joined the Escher Wyss - Oerlikon Company as a steam turbine engineer.

In 1961 he was employed by the Brown Boveri Company for their large Steam Turbine Department. Two years later, he became a chief design engineer of gears for turbomachinery and marine application. From 1969 to 1974, he was responsible for component standardization of the Brown Boveri - Sulzer Turbomachinery Ltd. He returned to Brown Boveri to join the Industrial Steam Turbine Control Department where he was promoted to his present position as a manager.



A native of Switzerland, Alfred Häusermann graduated as a mechanical engineer from the Lucerne State Institute of Technology. In 1966 he was employed by the Brown Boveri Company as a steam turbine engineer. He worked there for four years in the department for design and development of large steam turbines. In 1970 he joined the Turbine Testing Department. As a senior engineer, he was engaged in commissioning of large turbines and troubleshooting in steam power plants.

In 1975 he was promoted to a chief engineer and changed to the Industrial Steam Turbine Department. He is responsible for design and mechanical calculations of industrial and medium sized steam turbines.

ABSTRACT

The reliability of a steam turboset is dependent, among other things, on correct functioning of its hydraulic lubrication and control system.

An essential requirement is a reliable oil supply over the whole operating range. The main oil pump carries to a large degree this responsibility. The design and operational aspects of it will be dealt with in this paper.

Air dispersed in oil is known to endanger, in extreme cases, the safe and reliable operation of the turbogroup. Besides operating interferences of various system components an interruption of oil supply to the bearings may result in serious consequences. Without going into details of oil qualities, operational consequences as well as corrective measures, of air present in oil will be discussed.

The increasing specific demands imposed upon steam turbines may bring about new oil system difficulties. The subject matter presented here treats extreme cases of thermal loading of the lube oil as in the case when coming into contact with hot turbine parts. The resulting coke deposits in bearings may cause failure. With the assistance of experimental comparison tests lube oils on this basis are selected that will have the inherent capabilities to withstand such extreme operating conditions.

INTRODUCTION

In the following, several problems pertaining to the reliability of turbine oil systems will be discussed.

- Maintaining reliable oil supply over the whole operating range.
- Reducing the influence of the inevitable air trapped in oil to a safe operating minimum.
- Trouble free lubrication also in cases of high lube oil temperatures.

When working on such a problem subject it soon becomes evident that design and operational aspects in particular have priority. An attempt is made to present difficulties and their successful solutions based on actual operating experience.

HYDRAULIC CONTROL AND LUBE OIL SYSTEM

Systems built up as for example shown on Figure 1 have three main contributions to make towards operational safety:

- Control and lubrication during normal trouble-free operation.
- Protection of the turboset by reducing load or outright tripping when inadmissable malfunctions occur.
- Damage-free runout of the turboset after an emergency trip. This entails continuation of an adequate oil supply to the bearings.

It is quite clear that all system components contribute to the fulfillment of the above requirements: A closed hydraulic circuit has proved highly effective for the control (CS) and protection (PS) systems.

The constant pressure valve (CV) and the pressure reducing valve (RV) must be absolutely surge- and vibration-free in order to prevent fluctuations within the system. Experience has shown that simple spring-closed valves are not quite up to

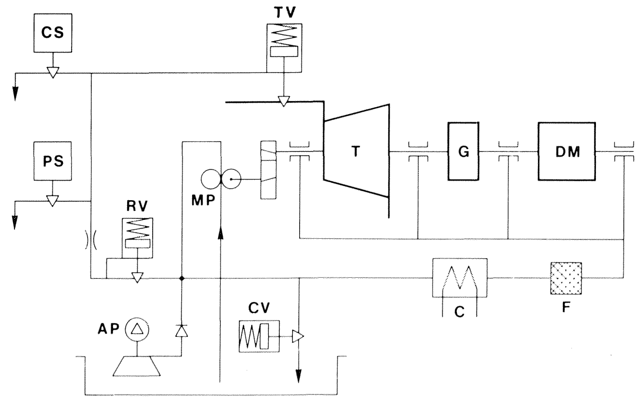


Figure 1. Schematic of Turbine Control and Lubricating System.

T	*Turbine*	*C*	*Cooler*
G	*Gear Box*	*CS*	*Control System*
DM	*Driven Motor*	*PS*	*Protection System*
MP	*Main Oil Pump*	*TV*	*Turbine Control Valve*
AP	*Auxiliary Oil Pump*	*CV*	*Constant Pressure Valve*
F	*Filter*	*RV*	*Pressure Reducing Valve*

the task in question. Quite often suitable dampening may help to solve the problem.

The main oil pump (MP) plays an important part in reaching the three main objectives mentioned at the beginning of this section. Oil flow must be guaranteed during all phases of operation. In a later section the relevant criteria and possible solutions will be discussed in detail.

As far as the filter (F) is concerned, the problems mostly center on filter quality, filter cleaning and cleaning intervals. The cooler's (C) main problem is the choice of heat exchanger tube material. In both cases a reliable change-over capability must be ensured when, for reasons of availability, two filters or coolers are installed in parallel, one being in service, the other on stand-by. Thereby an effective deaeration is of greatest importance; this problem will be discussed in the later section "air in oil."

RELIABLE OIL SUPPLY

Main Oil Pump Drive Variants

The reliability of a pump depends to a large extent on the drive method chosen.

Bearing failures of rotating machinery can occur when the oil supply is disrupted in some way. This can happen when an electric pump drive motor fails due to a power failure or malfunction of the protection system. A mechanical pump drive off the main shaft has the inherent advantage of uninterrupted oil flow for as long as the shaft is turning.

Mechanical drive of the pump is taken off the main shaft over a gear train with the appropriate drive ratios. Various arrangements are possible (see Figure 2).

The simplest case is (a). A single pump for lube and control oil pressurizes the system at 5 - 10 bar.

Case (b) comes into effect when pressure in the control oil system has to be at a different (higher) level. This pump is divided up into a high-pressure section delivering control oil at a pressure of 10 - 25 bar, and a low-pressure section for lube oil

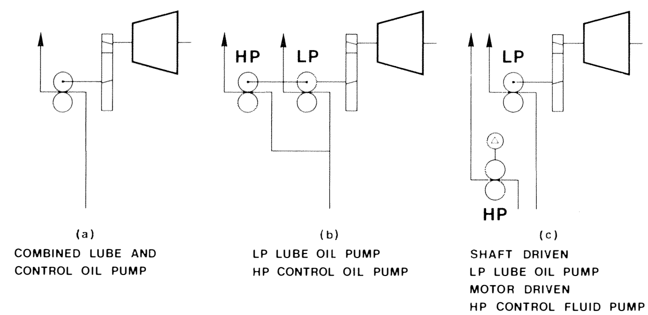


Figure 2. Schematic of Main Pump Drives for Lube and Control Oil Supply.

at 3-5 bar. A HP:LP delivery ratio of approximately 1:4 is considered most effective. If the turboset in question has a gear train with above normal oil requirements, an extra pump can be installed to cater especially for these.

The main priority lies in a reliable supply of low-pressure lube oil. The low-pressure pump should therefore be driven mechanically off the turbine shaft. Case (c) on Figure 2 shows another possible alternative with the high-pressure control oil pump driven by a separate electric motor. This variant has the advantage that the control oil pump can be shut down immediately in the event of a malfunction in the system.

A Failure-Free (Robust) Pump

Volumetric pumps are preferable as opposed to centrifugal pumps when used as main oil pumps. Oil flow is approximately proportional to shaft speed and supply pressure can be freely chosen within certain design limits. In contrast, the centrifugal pump has a speed-dependent pressure/flow characteristic which is variable and not always easy to fit to the requirements of a given system. Mechanically driven volumetric pumps can be designed as gear (helicoidal) or screw-type pumps.

Taking a gear pump as shown on Figure 3 as an example, the following are relevant for safety and reliability of operation: The teeth of the driving gears should be hardened and ground for long service life and to operate at the minimum noise level. The same goes for the pump gear pinions themselves.

The pump must be equipped with robust and durable bearings. Plain bearings with babbit or bronze bearing surfaces have proved most suitable. Experience has shown ball and roller bearings to have irregular service lives and to be unsuitable for use in main turbomachinery components. On the drive side the pump should be equipped with an especially tough bearing to take on the heavy radial and axial loads at this end.

Figure 4 shows a twin pump with separate high and low-pressure sections built according to the above design principals.

Pump's Suction Power and Noise Level

Apart from robust design and long service life the main oil pump must have a powerful suction capability and must operate within acceptable noise levels. Both requirements are limited by cavitation. Operational experience over many years and numerous tests have shown which methods can be employed to move the cavitation limit towards better suction and lower noise levels:

- Air content in the oil must be as low as possible, an acceptable level being less than 4% of volume at atmospheric pressure (1 bar).

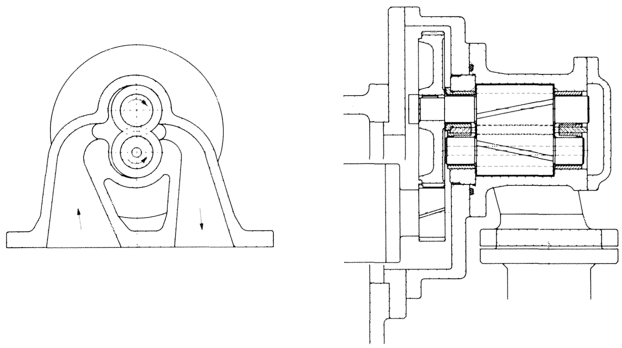


Figure 3. Gear Type Oil Pump Driven off the Turbine Main Shaft.

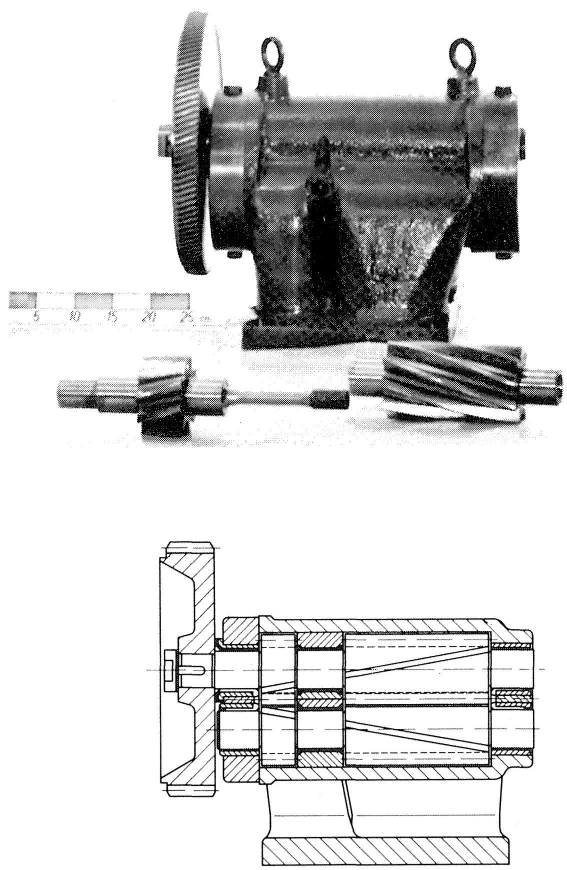


Figure 4. Gear Type Pump With Separated Sections for Lube and Control Oil

- Peripheral speed measured at the outside diameter of the pump gear pinions must not exceed 10-11 m/s. Noise levels are strongly influenced by the clearance between gear teeth and the inside of the pump casing. Small clearances lead to higher pumping efficiency but also raise noise levels considerably.
- The oil compression chamber formed by two gear teeth returning to the suction side of the pump must be pressure-relieved by slots in the sides of the casing [1].
- The suction chamber of the pump must be of optimum design. Figure 5 shows the stages in which the optimum shape II envolved from the originally simple shape I over an intermediate variant with very large suction chamber. The inlet channel is shaped so as to allow for gradual and complete filling of the gaps between the gear teeth.

Systematic evaluation of a number of test results have led to an extension of the caviation limit. Suction head (H) of the pumps incorporating the above-mentioned design improvements has been raised from level I to level II as shown on Figure 6. Sound pressure levels (S), measured 1 m away from

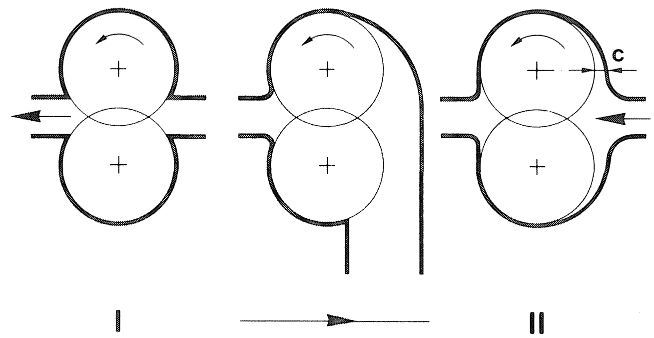


Figure 5. Improvement on Gear Pump Casing Shape.

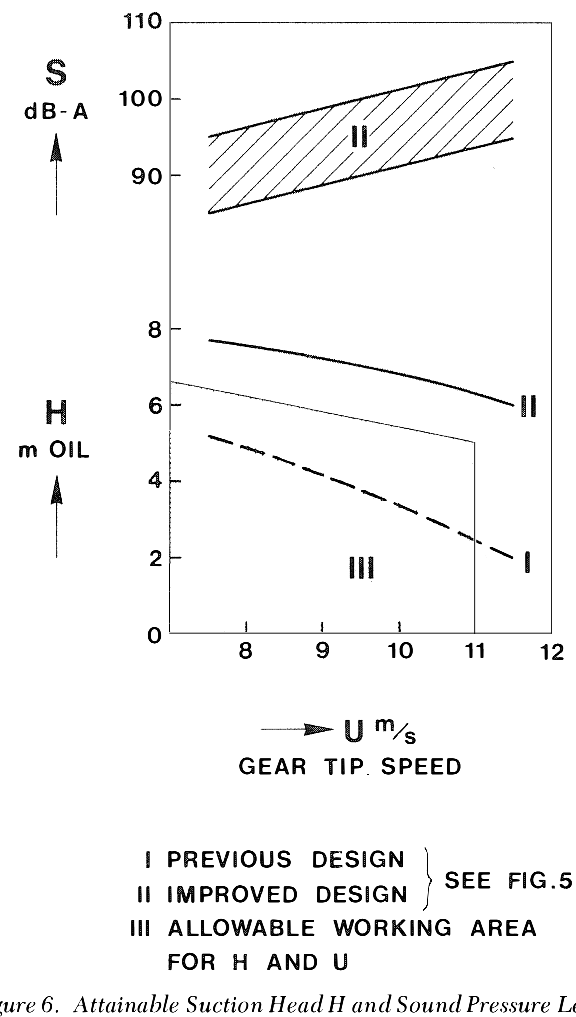


Figure 6. Attainable Suction Head H and Sound Pressure Level S for Gear Pumps.

the pump casing, are not overtly high when compared with the overall turbine noise levels. Area III denotes the H/U range for safe operation.

A pump with a good suction capability gets oil flowing very quickly. Supply only ceases when shaft speed sinks below 5% of maximum revs. The reliability bonus of a pump driven directly off the main turbine shaft is a result of this characteristic.

Incidents in a number of plants have shown that industrial turbines can run out without incurring bearing damage. This indicates that in the event of a turbine trip, when the auxiliary pumps are out of action due to malfunction or power failure, oil supply from the main oil pump suffices to guarantee a damage-free runout.

A mechanically driven main oil pump must also be designed to run backwards without incurring damage. Running backwards can occur in turbosets driving compressors or pumps when the medium to be transported flows in the reverse direction due to leaking or jammed valves.

AIR IN THE LUBRICATING OIL AND IN ITS SYSTEM

Air dissolved in oil and air-oil solutions are repeatedly known to have caused problems when used for lubricating purposes and hydraulic control functions of turbomachinery [2]. Problem areas are:

- Pressure surge in oil system
- Interruptions in oil supply
- Excessive formation of foam etc.

These negative effects of air present in oil can be eliminated to a great extent provided that the specific application requirements of the oil are observed and consideration be given to suitable design methods.

Following, a number of viewpoints on the problem cycle "air in oil" are given.

Type of Air in Oil

Figure 7 provides information on various types of air present in oil.

1. *Air Dissolved in Oil*

   Mineral oils, as well as all liquid mediums have the inherent capability to dissolve specific quantities of gas in molecular form without noticeable changes in their specific properties such as viscosity and compressibility. The mass of air taken up by the mineral oil in a dissolved state is solely a function of pressure and amounts, under atmospheric conditions, to approximately nine percent per volume. In theory, the potential air absorption ability is proportional to the pressure; that is to say, when pressure drops occur (e.g. at suction pump inlet) a certain quantity of the dissolved air volume is liberated. At a pressure increase, part of the air mass present in oil is again dissolved, however, a specific mass of air always continues to remain dispersed as bubbles in the oil. In general, air dissolved in oil does not influence its technical application for turbomachinery and can therefore in most cases be neglected.

2. *Air Bubbles in Oil*

   At the oil consumption points (e.g. bearings, control devices, gear boxes) oil is scattered and partly mixed with air. The discharged lube oil of the turbogroup is therefore diffused with air. The air is present in small bubbles of approximately 1/1000 to 1 mm and amounts to 2-10 percent by volume, giving rise to a considerable effect on viscosity and compressibility of the oil [3]. Oil condition (effect of additives, contamination etc.) is, besides the mechanical causes already mentioned, also responsible for high air content.

3. *Foam Formation on Oil Surface*

   A strong foam formation is, in addition to high air content, also based on the inherent tendency of the oil to produce foam [4]. Experience has shown that the machines are not endangered provided foam formation is kept within limits (foaming over of container).

4. *Air Plug in the System*

   In an oil system, large empty spaces such as is the case with standby oil coolers, standby pumps and standby filters, however not filled with oil, are functionally integrated in the cycle. If such devices are taken into operation without foregoing deaeration, the total air volume is transferred to the consumer. Due to the size of these air volumes, the formation of correspondingly large air plugs result. As a consequence, interrupted oil supply to the machines endangers its operation. The formation of air plugs will be discussed later in detail.

Effect of Air in Oil

With the following two examples, operational consequences of air in oil will be presented.

1. *Change-Over of Coolers and Filters*

   Oilcoolers and oilfilters are, in order to have 100% standby, often introduced as a duplicate parallel oil circuit into the lube oil system. The change-over from cooler one (filter) to cooler two (filter) is accomplished by means of coupled three-way valves located at inlet and outlet of coolers.

   Cooler and filter are emptied for cleaning and maintenance purposes. If upon recomissioning a sudden change-over, without prior deaeration, takes place, a large amount of air is introduced into the system. Figure 8 indicates the condition for that case in which the total pump output is assumed to be available for filling of coolers (filters). From this it is evident that if the oil supply is interrupted for several seconds or temporarily at least, an oil-air mixture is supplied. However, by way of a proper change-over procedure of parallel oil circuit coolers (filters) serious disturbances at bearings and hydraulic controls can be prevented. As shown in Figure 9, the cooler to be commissioned is filled up and thus oil pressure equalized by means of the filler valve and filler line F with simultaneous deaeration to oil tank. This filling and deaerating operation requires 5-20 minutes since nothing more than about 5-10 percent of the total oil supply is available. After switching the change-over valves (C) final deaeration for approx. 5 minutes is required.

   A much greater operating availability can be attained by having oil coolers connected in series since with this system the change-over and deaeration procedure is eliminated. The arrangement in series does not incorporate gate valves, oil flows through both coolers during normal operation. Each one of the coolers can however be cut off from the cooling water supply and emptied. The stand-by cooler plays no part in the actual cooling process and is brought into operation only when the cooler in service shows signs of malfunctioning. Cooling water supply to the dirty or eventually defective cooler is then cut off and cooler is emptied. Leaking tubes — if there are any — are sealed for the time being, with final repairs to take place during major shut down.

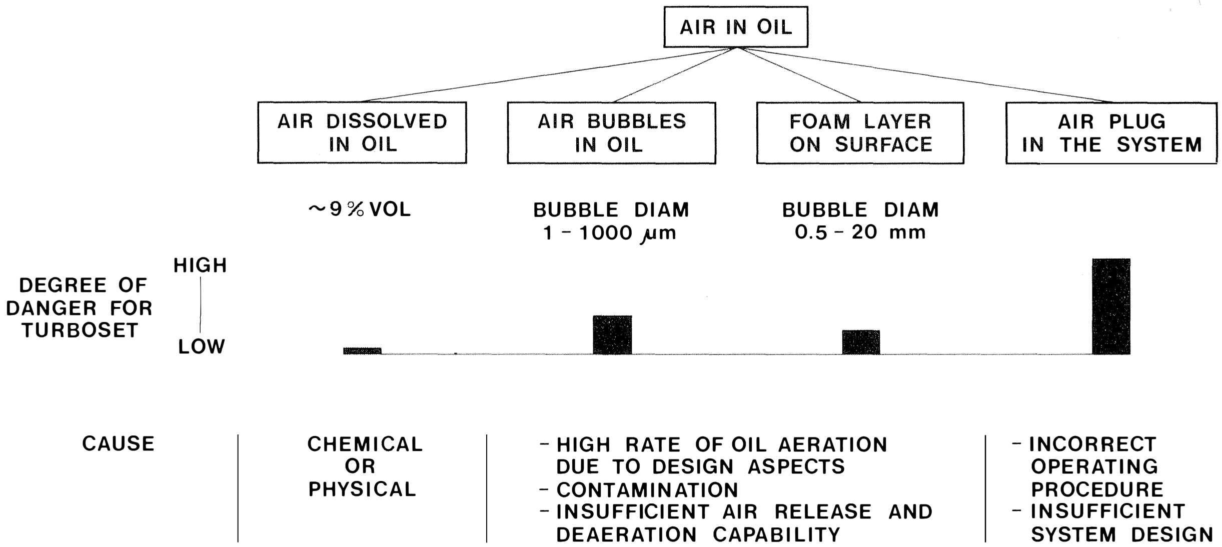
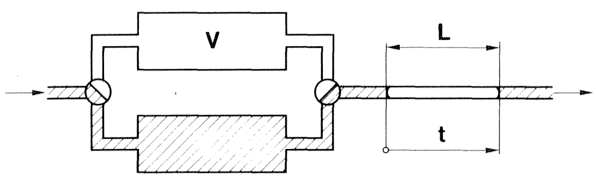


Figure 7. Different Kinds of Air in Oil.



			COOLER	FILTER
VOLUME	V	m^3	0.2 ÷ 0.4	0.05 ÷ 0.1
AIR PLUG	L	m	15 ÷ 30	4 ÷ 8
INTERRUPTION OF OIL SUPPLY	t	s	10 ÷ 20	3 ÷ 6
OR				
SUPPLY OF 50 % OIL/AIR	t	s	20 ÷ 40	6 ÷ 12

Figure 8. Interruption of Oil Supply Due to Instant Cooler or Filter Change-Over Without Deaeration.

2. *Deaeration of Submerged Pumps*

In order to secure a continuous oil supply several pumps are provided in the system with circuit separation by means of non-return valves.

Figure 10 shows that space (R) between pump and non-return valve may, during a standstill, run partly empty. By the application of a breather line air discharged into the oil-system, during start-up of this pump, can thus be prevented. Venting must take place at point (D) just above tank level.

According to the model shown in Figure 10 difficulties that have been encountered, may be explained if venting the system takes place below the oil level. Rising air bubbles will force oil level in space (R) downwards until eventually oil level (h) is lower than the pump suction level. Finally the pump suction action ceases, although properly mounted below the oil level.

Air Release

By applying suitable design and operational measures it is possible to reduce air penetration into the oil system thus reducing air absorption by the oil. When selecting oils emphasis must also be placed upon its quality to have good air separation combined with low dispersion. To be considered as well are the chemical and physical properties of the oil since, for example, a low viscosity oil deaerates much better than a high viscosity oil thus a warmer oil deaerates better than a colder oil. Because of this, cooling should actually take place after leaving the oil tank.

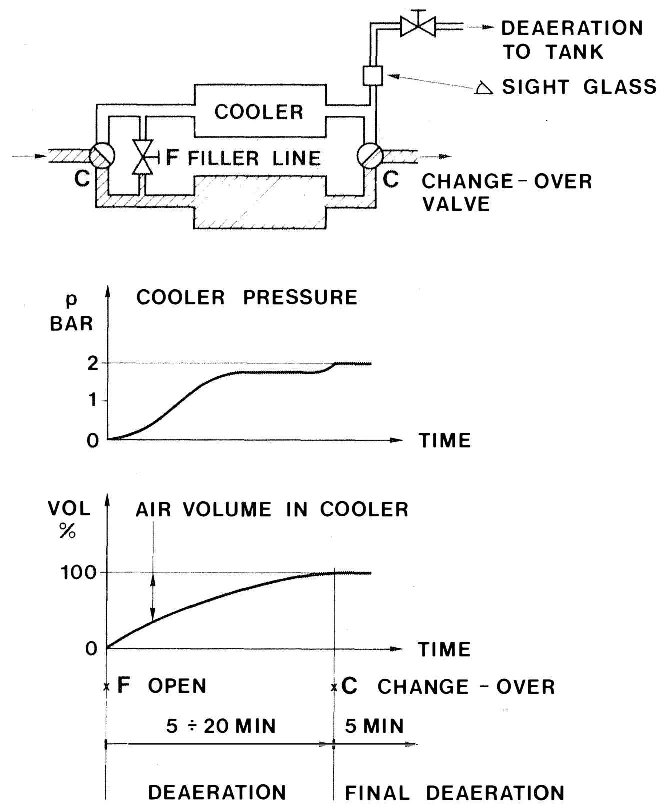


Figure 9. Oil Cooler Change-Over.

The oil must have the opportunity to give off air. During a period of stabilization in the oil tank air bubbles are given a chance to rise to the surface and thus separate from the oil. Experience in design and operation of turbomachinery has shown that a retention time of approximately 6 to 10 minutes provides sufficient deaeration of the oil.

Figure 11 indicates results from a deaeration test taken with an oil tank based on a 5 minute oil retention time. The retention time is of great importance for oil deaeration. By introducing baffles as well as guides the deaeration effect has been improved from A to C. The essential objective of introducing baffles is based on the fact that by this design measure the total tank volume does actually take part in the process of stabilization.

The maximum rate of deaeration has been obtained during test (D), by treating the oil with an ultrasonic generator. It is possible to reduce the air content in the oil to practically zero within two minutes. This solution is conceivable in cases of high air retention.

THERMAL LOADING OF LUBE OIL

Over a number of years now, the lube oil characteristics specified and tested to guarantee trouble-free lubrication have not changed. Due to the increase in turbo-machinery and gearbox performance one of the familiar problems has been aggravated:

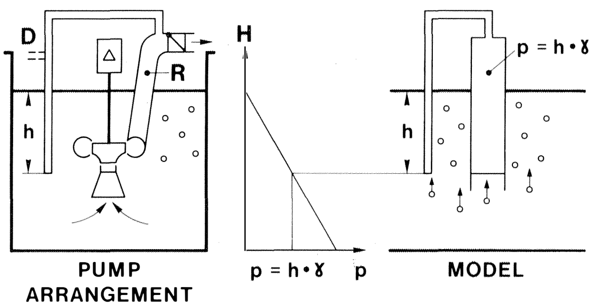


Figure 10. Deaeration of Submerged Pumps.

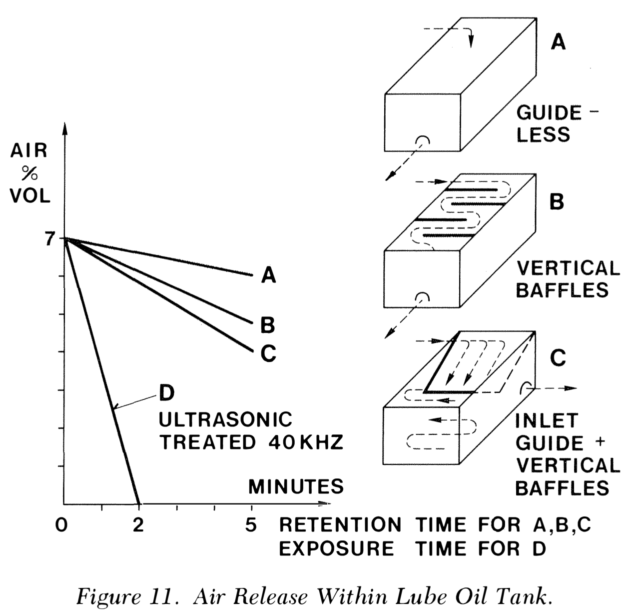


Figure 11. Air Release Within Lube Oil Tank.

When lube oil, on its path through the machinery, comes into contact with hot spots, it undergoes thermal loading which can have consequences for the oil and certain parts of the lubricated machinery.

Thermal Loading and its Consequences

Lube oil itself ages quickly and to a considerable degree due to thermal loading. Hot spots sprayed with oil can experience surface changes as described below for the example of a turbine and gearbox rotor.

Figure 12 attempts to show the various possible consequences diagramatically. Lube oils of different makes, otherwise equally proficient at satisfying the demands placed on them, react differently to thermal loading and show varying tendencies to oil lacquer formation at high temperatures.

At medium metal temperatures up to approximately 120°C (250°F) the hot spots in question can experience discoloration through oil lacquer formation, the color changing to golden yellow and as far as dark brown. This discoloration, however, causes no operational problems.

At higher metal temperatures, depending on the oil, over 100-140°C (212-285°F), the oil lacquer deposits become carbonised coke-like deposits (5) which can cause substantial operational problems as will now be discussed:

A. Extensive carbon deposit formation on the hot shoulder (S) of the turbine rotor. Together with lube oil these deposits can form an abrasive layer which in turn can lead to substantial rotor wear (W).

B. Oil lacquer deposits on gear teeth are normally ground off by tooth interaction and settle inside the V of the gaps between gear teeth. Large deposits can cause pressure on the corresponding gear tooth tip and thereby press the gear wheels apart.

C. Experience with an actual case has shown that the build-up of coke deposits in plain bearings is especially

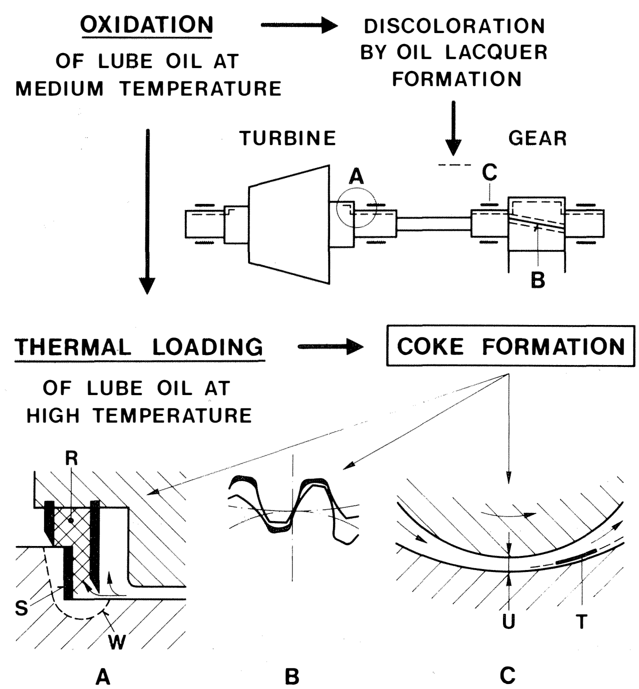


Figure 12. Consequences of Thermal Loading of Lube Oil.

dangerous. At the point of highest bearing temperature (T) immediately following the narrowest portion of the lubrication gap (U) the formation of deposits narrows the lube gap still further and can thus cause bearing failure.

Figure 13 shows a gear sprocket bearing at an early stage of failure; at a later point in time the white metal melted. This caused the gear wheel to run out of true and led to gear tooth failure.

Counter Measures

This situation can be countered with design measures on the one hand and the use of thermally stable lube oils on the other. The following are possible measures on the design side:

- The rotor section between the bearings and the hot labyrinth section of the turbine rotor must not be built too close together so as to keep the heat as far away from the oil flooded bearing housing as possible.
- The oil path through the machinery must be chosen in such a way as to keep the oil away from potential hot spots.
- In extreme cases a special cooling system for hot running components such as bearings and gear teeth must be envisaged.

The lube oil must be as resistant as possible against thermal loading. The Diesel test according to WOLF (DIN 51392) has proved to be an acceptable method of testing this capability. The lube oil is routed over a heated metal strip, as shown on Figure 14 and the formation of deposits monitored. The amount of the deposits, the neutralization value and the viscosity change are measured. In an evaluation diagram, Figure 15, the measured parameters are triangulated and points marked on the graph for different oils. The gear sprocket bearing mentioned above failed while running on oil No. I. When oil No. II was used the bearing ran trouble-free without any deposit formation.

The tests so far have shown that the WOLF test does not give an absolutely clear picture of the oil's characteristics. The best results can be achieved by comparing the visual aspect of the test strip, the results as shown on Figure 15 and operational experience for various oils. In this way and by continuously refining the existing test methods or designing new ones it will be possible to prevent failure even under extreme conditions.

CONCLUSIONS

The foregoing examples have illustrated that within an oil cycle permanent problems exist for instance in substaining the oil supply, air in oil, etc. Additionally, new difficulties may arise due to thermal overload of the oil caused by excessive operational demands of turbosets.

Experience has shown that besides continuous efforts to strive for suitable oil qualities it is equally important to pay specific attention to the design of the system. With an appropriate construction of the oil-system and its components, based on knowledge gained by test and supported by experience a trouble free operation should be attained.

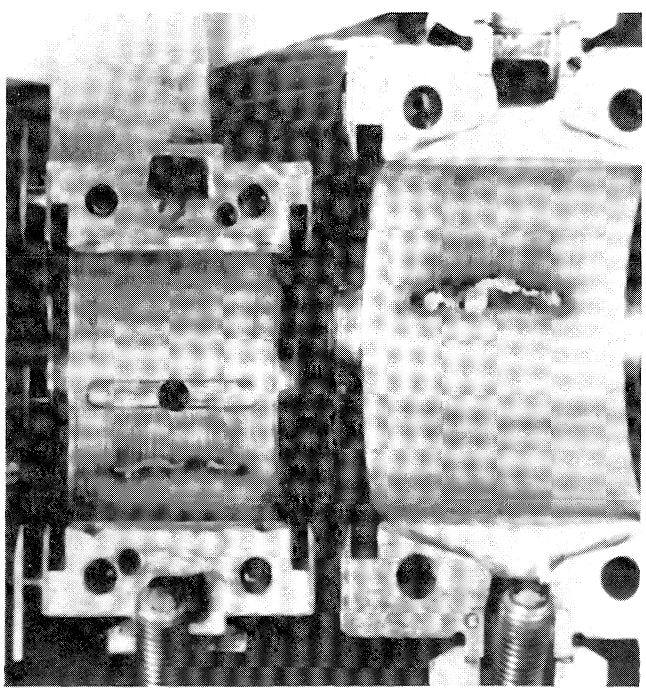


Figure 13. Build-up of Coke Deposits in a Bearing.

Journal Diam.	*125 mm (5 in)*
Speed	*9600 rpm*
Spec. Loading	*300N/cm² (425 psi)*
Max. Metal Temp.	*107°C (225 °F)*

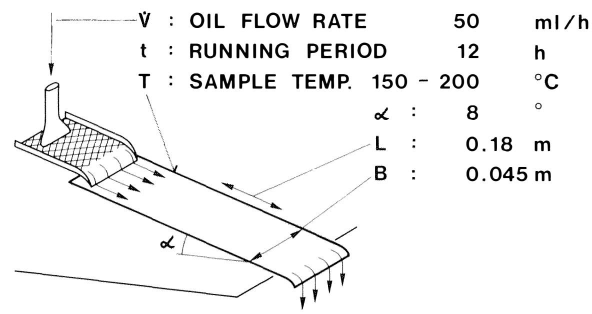


Figure 14. Thermal Stability of Lube Oils, "WOLF" Strip Test.

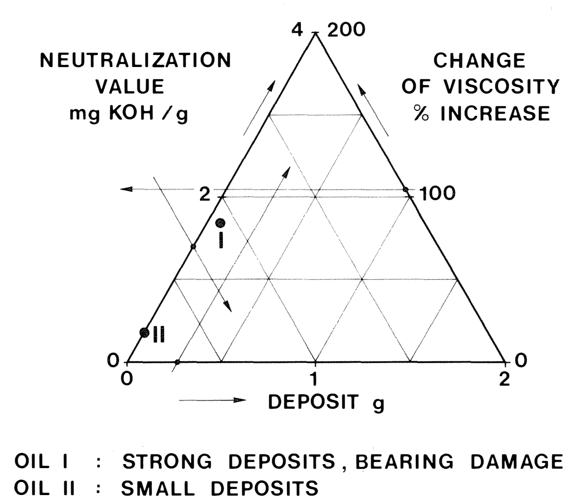


Figure 15. Thermal Stability of Lube Oils, Examination of WOLF Test Results.

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